THE ENGINEER

MARCH 16, 1956

Vol. 201 No. 5225

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A.C. Electrification on British Railways

Two important decisions about the £1200 million modernisation plan for British Railways were announced last week by Sir Brian Robertson, chairman of the British Transport Commission. The first is that the standard for future electrification will be alternating current at 25kV and 50 cycles per second; the second—a strange one which we hope to discuss in a later issue—is that the vacuum continuous brake will be fitted on all freight rolling stock. In choosing the high-voltage mains-frequency a.c. system in preference to the well-tried d.c. system the British Transport Commission has wisely decided to forget the past and to invest for the future, which, as we suggested in a leading article on February 10th., is particularly bright for the new method of electrification. The choice, once made becomes "virtually irrevocable" but that it is well founded is confirmed by the results of a comparative survey of the London Midland Region main route from Euston to Manchester and Liverpool—a route on which the advantages of a.c. are less marked than on other lines. The comparison is illuminating, for it shows that the capital costs would be £117·8 million for 25kV a.c. compared with £123·6 million for 1500V d.c. Allowing for the provision of clearances the cost of power supply and overhead line would be cheaper by £9·3 million with a.c., which would more than compensate for the fact that the motive power costs would be £2·3 million higher. Moreover the latter figure makes no allowance for the reductions that may be expected as locomotive building gets into its stride. Finally the estimates allow a pessimistic figure for contingencies, of £6·1 million for a.c. compared with £4·4 million for d.c. Because of the heavy traffic the motive power requirements are correspondingly heavy and represent a greater fraction of the total cost than would be the case on other lines. On no other main route would the cost of providing electrical clearances be so high as between Euston, Manchester and Liverpool. By using a.c. instead of d.c. the saving in capital cost per mile would, therefore, be even greater on other lines to be electrified under the modernisation plan. On the Euston-Manchester-Liverpool scheme a reduction of £1,000,000 in annual costs is anticipated by the adoption of a.c. instead of d.c. and corresponding economies are to be expected on other lines.

Some interesting details emerge from the comparative survey. A.C. electrification would require only 12 transformer substations, which would be owned by the Central Electricity Authority and, because of the proximity of existing C.E.A. supply points, only 17 miles of new feeders would be needed; d.c. electrification, on the other hand would require 70 rectifier substations and 250 miles of 33kV cable. Without losing sight of the possibilities of developments in the application of the a.c. series motor British Railways propose to concentrate for the present on the rectifier locomotive with d.c. series motors and it was this locomotive that was compared with the straight d.c. locomotive in the survey. Because of the superior adhesion, weight-for-weight, of the a.c. locomotive with d.c. motors, lighter locomotives could be used for a given duty and the reduced wear and tear on the track would react favourably on the overall cost of the modernisation plan.

The Manchester-Crewe section will be the first stage of the electrification and will serve as a proving ground for new equipment. The new system will be used for 1210 of the 1460 route miles which are to be electrified on British Railways. The main trunk lines involve a total of 820 route miles: Euston, Manchester, Liverpool; Kings Cross, Doncaster, Leeds and possibly, York; and Liverpool Street, Ipswich, in extension of the existing electrification which will be converted from 1500V d.c. to 6·6kV a.c. Similarly a.c. will be used for suburban electrification (excluding extensions on the Southern Region) totalling 390 miles: London Tilbury and Southend; Liverpool Street, Enfield and Chingford; Liverpool Street, Hertford and Bishop's Stortford; Kings Cross, and Moorgate to Hitchin and Letchworth, including the Hertford loop.

A Well-Timed Ceremony

Every now and then a little, almost domestic, ceremony, planned to be only of local importance, attains a much broader significance because of some accident of timing. Last autumn W. H. Allen Sons and Co., Ltd. of Bedford presented certain machine shop equipment to Bedford School, a school in which the company is naturally particularly interested. It was suggested at the time that a plaque might be affixed to a wall of one of the school's workshops to record the gift. And so it was done. Suitable wording was devised, a suitable bronze plaque was made, a domestic unveiling ceremony was planned and a date was fixed. Under ordinary circumstances the event might have attracted no more than a bare mention in our columns. In fact, however, the unveiling ceremony, held last Friday, followed closely upon the publication of the White Paper on Technical Education, a Paper which stressed the need of this country to improve and expand its facilities for scientific education in schools, as well as colleges, and which preached throughout the imperative necessity for an expansion of the country's output of technologists. How succinctly the wording on the plaque at Bedford School stresses the same point! It was presented "to ensure that boys passing through Bedford School shall be continually reminded of the absolute dependence of Britain on engineering as a profession". It was peculiarly appropriate and it gave added significance to the event that the plaque was unveiled by the President of the Institution of Mechanical Engineers, Mr. P. L. Jones. For by his presence he signified the recent growth of the interest of the major engineering institutions in the schools of this country additional to that which they have always shown in colleges and universities providing a more specialised training in engineering. The whole affair in fact stressed how important it has become, if a supply of technologists adequate to this country's needs is to be secured, that the interest of boys, still at school and still with their future profession unchosen, should be awakened to the fascinating possibilities that a career in engineering can open before them.

Aero-Engine Test Plant

Some details are now known of the high altitude test plant which is being built by Rolls-Royce Ltd., on a site adjoining its Aero Engine Division at Derby. The main compressor and exhauster station will contain an axial compressor of 157,000 cu. ft. per min. capacity and five other compressors. They will operate two engine test cells, two combustion test rigs, a supersonic wind tunnel and a compressor test rig. The high altitude test chambers, one for jet and the other for propeller turbines, will be approximately 75 ft. long and 12 ft. 6 in. diameter, constructed of nickel alloy steel. The air enters through refrigerating and drying equipment and can be supplied at a wide range of pressures and temperatures, as for instance those experienced at an altitude 70,000 ft. and a speed of 1,500 m.p.h. From a plenum chamber the air passes through a honeycomb straightener and a mesh guard directly to the engine intake, with a bleed to the chamber itself. The exhaust gases are cooled in two stages to 40 deg. cent before the compressors return them to the atmosphere. Each of the combustion test cells will supply a full sized annular combustion chamber, one running above and the other below ambient pressure. Kerosene-burning pre-heaters offer inlet temperatures up to 450°C. The wind tunnel will run up to Mach number 4 and have a working section approximately 3 ft. square. A 15,000 h.p. air turbine will drive the compressor test rig. About 80 MW at 33 kV will be taken from the Grid, and cooling towers will be provided to reduce water consumption. Calibration of the high altitude test beds should be completed by the end of 1957.

World Speed Record of 1,132 m.p.h.

A new world speed record of 1,132 m.p.h., subject to confirmation by the Federation Aeronautique Internationale, was set up on March 10th in a Fairey Delta 2 piloted by Mr. Peter Twiss. Two flights were made over a nine mile course between Chichester and Ford, Sussex; the first was at 1,117 m.p.h. and the second at 1,147 m.p.h. The speeds were measured on photographic equipment which was designed, made and operated by staff of the Royal Aircraft Establishment, Farnborough. The Fairey Delta 2, which was ordered by the Ministry of Supply for research into the problems of supersonic flight, thus becomes the first aeroplane to exceed a speed of 1000 m.p.h. in straight and level flight, over a measured course, in conditions recognised by the Federation Aeronatique Internationale for a world speed record. This aircraft, illustrated in our issue of January 6th last, has thin delta wings with 60 degree sweep-back on the leading edge and a needle nose that can be lowered for improved view in taxying and landing. The power unit is a Rolls-Royce Avon jet engine with re-heat.

Letter to the Editor

(We do not hold ourselves responsible for the opinions of our correspondents)

THEORY OF HYDRODYNAMIC LUBRICATION IN PARALLEL SLIDING

SIR,—The article by W. Lewicki, appearing in your issue of December 30, 1955, revealed an interesting new approach to the age-old problem of parallel sliding. Many experiments which I had carried out (to recommend materials and lubricants for aero-engines) led me to the belief that some very powerful shear-vortex forms at the leading edge of sliding members such as piston rings, plain thrust bearings and the like. I should like to mention just one of these experiments.

Fig. 1 represents a rotatable housing or chuck A carrying four test pieces P, the

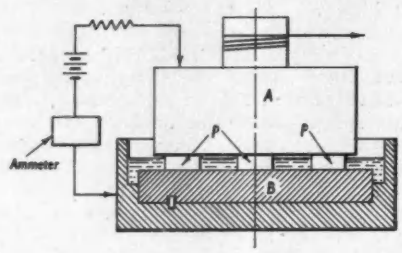


Fig. 1—Apparatus for starting sliding pads from rest

surfaces of which were carefully lapped to be smooth and coplanar. These rested on the smooth, hard surface of a steel disc B, the contact pressure between the two being about 20 lb per square inch. The assembly was enclosed within a heated oil bath. The rotatable member A was given a uniform acceleration from rest. A steady current flowed through the contact surface, as indicated on the ammeter, until a certain speed was attained, when the current became suddenly interrupted and remained so for all higher speeds. The speed of current interruption, presumably the speed at which a hydrodynamic film formed, was found to increase with increase in oil bath temperature, as shown in Fig. 2.

It is seen that the formation of a fluid film between parallel surfaces depends (1) on the

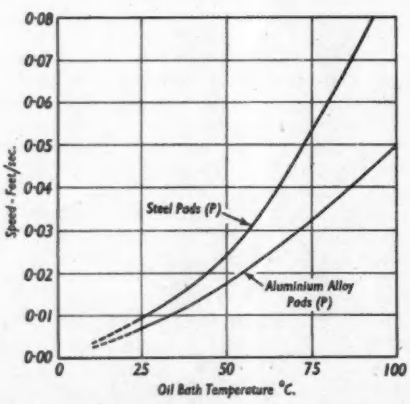


Fig. 2—Formation of film due to pads starting to move

relative speed between the surfaces, (2) on the viscosity of the lubricant, and (3) on the nature of the test pieces, i.e. whether made of steel or aluminium.

It seems to me that the presence of excess pressure at the leading edges, to feed lubricant into the sliding surfaces (and then to jack them up) is a perquisite of the whole thing.

An investigation into the shape of the leading edges, ascribing the pressures to the wedge formed by various chamfers, led nowhere. It was then assumed that the leading edge formed a strong source originating pressure, of mechanism unknown, and a linear pressure gradient was considered to extend along the length of the slider. This, together with a few expressions relating thereto, is indicated in Fig. 3, and formed the basis of comparison between the behaviour of various materials under sliding conditions, most of which adhered approximately to the law

$$\mu = K\sqrt{\frac{\eta U}{W}},$$

where

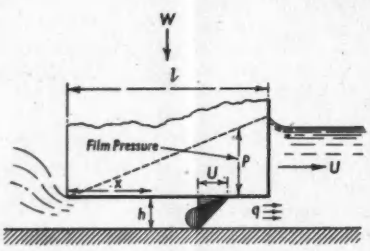
μ is the coefficient of friction,
η is the oil viscosity,
W is the total load on the sliding surfaces unit width,
U is the relative velocity of the system,

and

K is an empirical constant depending on the materials, smoothness of the surfaces and properties of lubricants apart from viscosity.

As a matter of interest if we assume that the passage of the slider over the lubricated disc has a minimum effect in disturbing the oil, i.e. the flux through the film is zero (flow due to the pressure gradient balancing out the flow due to the shear, $Uh/2$), we find that $K = 4/\sqrt{3}$. Other assumptions about the flow through the film will give other values for K.

Fig. 4 shows some results of experiments which show that the coefficient of friction in



Assume pad of length l advances at U.

Total flux in film

$$q = \frac{Uh}{2} - \frac{h^3}{12\eta}\frac{dp}{dx}.$$

Put $q = 0$,

then $$\frac{dp}{dx} = \frac{6\eta U}{h^3}.$$

$$W = \frac{3\eta U l^2}{h^3} = \text{Load per unit width.}$$

$$\mu = \frac{4}{3}\cdot\frac{h}{l} = \frac{4}{\sqrt{3}}\sqrt{\frac{\eta U}{W}}.$$

Fig. 3—Slider with highest film pressure at leading edge and zero film flux

parallel sliding obeys laws of this nature over a considerable range. This range has been shown to be one in which a continuous lubricant film exists between the surfaces, the evidence in this case relying on certain electrical measurements described elsewhere.

Film breakdown is accompanied by a rise in coefficient of friction, conditions becoming more severe as we move to the left. Tests show that the lubricating film begins to break down, and, of course, one would not then expect the simple law to be obeyed any longer.

When running cast iron sliders against a cast iron disc, the value of K is 1·5. The fluid film law

$$\mu = 1{\cdot}5\sqrt{\frac{\eta U}{W}},$$

was found to hold for values of

$$\sqrt{\frac{\eta U}{W}} \text{ exceeding } 1{\cdot}5\times10^{-3}.$$

Phosphor-bronze pads against hard nitrided steel gave higher values of K, namely, 4·0,

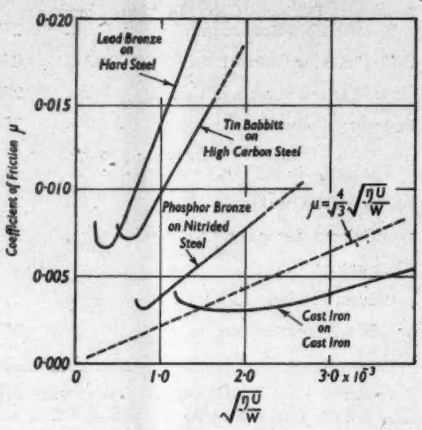


Fig. 4—Coefficient of friction in parallel sliding

but operation of the sliders could be carried out at much more severe conditions, the minimum value of the variable being of the order $0{\cdot}85\times10^{-3}$. Thus three times the load could be carried by this pair before the film broke down. In actual fact loads of the order of 1 ton could be carried by a 3·5in diameter thrust bearing running at about 600 r.p.m., the latter consisting of only three or four isolated pads.

The problem of such parallel slider bearings is not that of developing sufficient film pressure to sustain the load, but to remove the heat sufficiently rapidly from the pads.

Kerosene, petrol and other liquids can be used to lubricate parallel surfaces under this régime, the only difference between their performance and that of oil being due to their lower viscosities.

I should like to conclude with one or two observations about the behaviour under parallel sliding. The action does not appear to be due to the Rayleigh effect, which presupposes a cyclic variation in film thickness of about 3 : 1. It was, indeed, found that each pad had to project from its housing (or, the inter-pad grooving be machined down) by at least 0·060in. Assuming the film thickness of the order of 10^{-4} in, the ratio of depth of lubricant between pads to that under the pads appears to be in the hundreds. The depth of the leading edge, in fact, should be many orders of magnitude greater than the pad film thickness.

The question of spacing of the pads is important, too. Sliding pads should be separated from each other in the direction of motion by distances at least equal to their length, the separation increasing with linear speed of operation.

Equation (19) in W. Lewicki's article implies that the load-carrying ability of a slider depends directly on the conditions in the leading edge vortex, the film under the slider conforming to the conditions at the leading edge. In the experiments I have

mentioned the load-carrying ability of the sliders (sliders of less than 1 square inch area carried many hundreds of pounds load each) seemed to be more specifically related to laws which imply a high-pressure source at the leading edge merely sufficient to feed lubricant into the film space to balance a given load. No limitations of such pressure were evident. There always seems enough !

It might appear therefore that the type of "shear vortex" described by W. Lewicki is more powerful than suggested at present by the mathematics. Further investigations on the lines already pursued will be followed with great interest. Perhaps they will at last clear up the uncertainty which still persists as to why accurately machined components slide together without seizing up.

V. H. BRIX

Saughall, Chester.

Literature

Flood Estimation and Control. By B. D. RICHARDS, M.I.C.E. London : Chapman and Hall, Ltd., 37, Essex Street, W.C.2. Price 35s.

THE third edition of this well-known work includes some additional information on various aspects of flood estimation. The "unit hydrograph" method is discussed more fully, and the adaptation of the author's method of flood estimation to the special case of a reservoir of large area in relation to the catchment is discussed. A further addition to the author's theory relates to large catchments with a long period of concentration, and a method of constructing a hydrograph of a flood caused by a moving storm is developed. The author's method of approximate calculation for flood absorption effect of a reservoir is set out more fully in this edition.

Recent information on floods is also included in the new edition, and several of the tables of data contain recent records. The disastrous flood at Lynmouth in August, 1952, is discussed, and an estimate is made of the intensity of this flood and of those of maximum normal and maximum catastrophic floods from the catchment.

Plant Engineer's Easy Problem Solver. By LOUIS J. MURPHY. London : Arthur F. Bird, 66, Chandos Place, W.C.2. Price 52s. 6d.

THIS is a book of nomographs and charts designed by the author to provide a quick and rapid means of solving problems of many kinds commonly encountered in engineering practice. Each chart is accompanied by a concise explanation and an example showing how it is to be used. The range of subjects covered by the book include problems in metal working ; heating, ventilating and air conditioning ; materials handling ; compressed air ; pipes and piping ; mechanical power transmission ; refrigeration ; pumps and pumping ; engines ; plant maintenance, &c.

Books Received

Neutron Cross Sections. U.S. Atomic Energy Commission. London : McGraw-Hill Publishing Company, Ltd., McGraw-Hill House, 95, Farringdon Street, E.C.4. Price 90s.

Control of Nuclear Reactors and Power Plants. By M. A. Schultz. London : McGraw-Hill Publishing Company, Ltd., McGraw-Hill House, 95, Farringdon Street, E.C.4. Price 52s. 6d.

The Master Diver and Underwater Sportsman. By T. A. Hampton. Adlard Coles, Ltd., 7, Brunswick Place, Southampton. Price 15s.

Oil Flow and Film Extent in Complete Journal Bearings

By J. A. COLE, M.Sc. A.M.I.Mech.E,* and C. J. HUGHES†
No. 1

This paper was read before the Institution of Mechanical Engineers on February 10th and is here abstracted.

In 360 deg. journal-bearing theory, the load-carrying film is assumed to be partial as a result of cavitation in the diverging outlet film, but the friction producing film is taken as complete. The effect of oil inlet conditions on film extent and the state of affairs prevailing in the cavitated film have been matters for speculation. A method has been developed for photographing the oil film in glass bearings of standard clearance ratio (0·001 and 0·002) and the results of a qualitative and quantitative examination of film extent are described. The measured oil outflow rates are analysed into the hydrodynamic and supply-pressure-induced components, and explanations for an apparent deficit in the former are discussed.

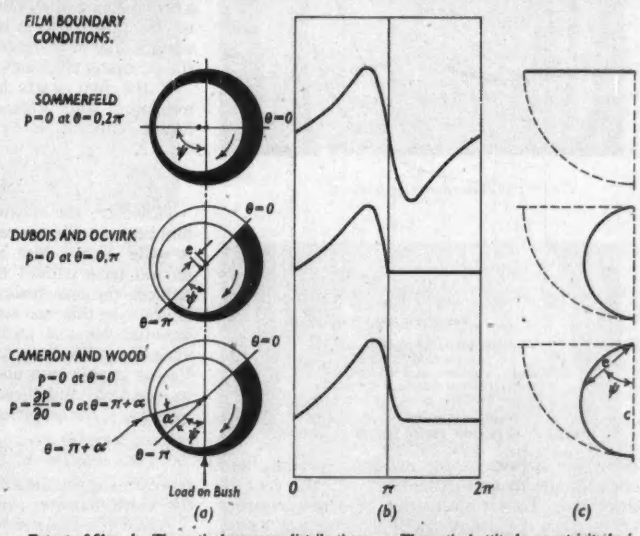
IN hydrodynamic lubrication theory, the performance of the plain journal bearing is derived from the solution of the Reynolds equation for viscous flow in a thin lubricant film :

$$\frac{1}{r^2}\cdot\frac{\partial}{\partial\theta}\left(\frac{h^3}{\eta}\frac{\partial p}{\partial\theta}\right)+\frac{\partial}{\partial z}\left(\frac{h^3}{\eta}\frac{\partial p}{\partial z}\right)=\frac{6U}{r}\cdot\frac{\partial h}{\partial\theta} \quad . \quad (1)$$

The process involves assumptions regarding the film boundary conditions, and in the case of the complete (360 deg.) journal bearing the form and extent of the oil film have long been a matter of controversy.

One of the more satisfactory solutions is that obtained by Cameron and Wood (1949),* using Southwell's relaxation method, and later extended by Sassenfeld and Walther (1954). Here it is assumed that pressure is generated over only part of the bearing circumference, but that friction loss occurs by the shearing of a complete and continuous oil film. The arrangements for oil supply to the bearing are not considered, and the side-leakage oil flow is determined only by the film pressure distribution deduced.

The study of film extent described in this paper arose from observations that measured friction torque was lower than that calculated on the assumption of a complete film. Since there is also uncertainty in the value of the representative film viscosity to be used in this calculation an investigation of bearing temperature-distribution has also been undertaken, but this will be described at a later date.



a—Extent of film. *b*—Theoretical pressure distributions. *c*—Theoretical attitude-eccentricity loci.

Fig. 1—Extent and distributions of pressure films

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Notation.

b—Bearing width.
c—Bearing radial clearance.
d—Bearing diameter.
e—Bearing eccentricity (relative displacement of bush and journal centres).
h—Film thickness (subscript indicates position).
N—Shaft rotational speed.
P—Bearing load per unit projected area.
p—Film pressure at a point, oil supply pressure.
Q—Lubricant flow rate (without subscript, side leakage flow, with subscript, circumferential flow at position indicated by subscript).
r—Bearing radius.
S—Sommerfeld number, $\frac{\text{viscosity } \eta \times \text{rotational speed } N}{\text{load pressure } P} \times \frac{1}{(\text{clearance ratio})^2}$
U—Surface speed.
x—Co-ordinate in direction of bearing width.
ε—Eccentricity ratio *e*/*c*.
η—Viscosity.
θ—Angular co-ordinate (origin taken at position of maximum film thickness).
$\overline{\pi+\alpha}$—Angular co-ordinate of film end, when $p=\partial p/\partial\theta=0$.
ψ—Attitude angle (angle between load line and line of centres).

THE EXTENT OF THE LOAD-CARRYING FILM

The simplest assumption, that the film is continuous (from 0 to 2π in Fig. 1, angles being measured from the position of maximum film thickness), leads to the classical Sommerfeld solution for an infinitely wide bearing, the second term of equation (1) being considered negligible compared with the first. The result predicts an attitude angle of 90 deg. and a pressure distribution skew symmetrical about the pressure datum. At any appreciable bearing eccentricity, negative pressures are then liable to occur on the unloaded side of the bearing unless oil is admitted either at a considerable pressure or at the variable position of the minimum film pressure, arrangements which are normally impracticable. A continuous film was assumed by Muskat and Morgan (1938) in their analyses of the complete journal bearing.

To avoid the negative pressure difficulty, the pressures in the unloaded half of the bearing may be neglected by assuming that cavitation occurs there, and the limits of the load-carrying film are then taken as 0 and π. This is mathematically convenient, but it implies a flow discontinuity at the film end. These boundary conditions were used by DuBois and Ocvirk (1953) when developing the approximate theory for the infinitely narrow bearing, in which the first term of equation (1) is considered to be negligible compared with the second, and by others.

More realistic load capacities are obtained on disregarding the contribution of the unloaded half of the bearing, and the straight line attitude-eccentricity locus is replaced by a curve generally similar to the roughly semicircular locus observed in practice.

The currently accepted boundary conditions are that the positive pressure region of the film starts at $\theta=0$ and ends at $\overline{\pi+\alpha}$, where $\left(\frac{\partial p}{\partial\theta}\right)_{\overline{\pi+\alpha}}=p_{\overline{\pi+\alpha}}=0$. The first explicit mention of them is by Gümbel and Everling (1925), and they have been used by Cameron and Wood (1949), and Sassenfeld and Walther (1954), in their analyses of complete bearings of finite width. The second condition avoids the flow discontinuity already mentioned, and is compatible with shaft stability, questions discussed by Swift (1931), Cameron and Wood, and others.

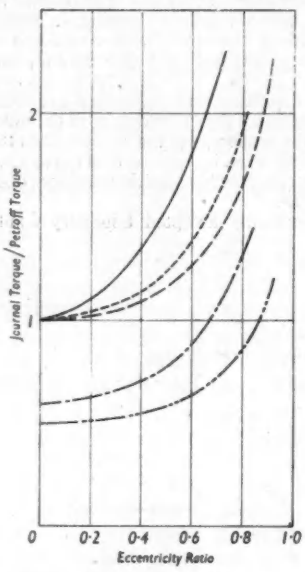
Pressure plotting has been used by a number of investigators to determine the extent of the load-carrying film in complete bearings. Stanton (1922), using large clearance ratios and low oil-feed rates, obtained very short pressure films, but otherwise results do not contradict the 0 to $\overline{\pi+\alpha}$ film extent. However, near the film ends, where the pressure measurements are least accurate, there is some doubt, the observed pressure distribution being flatter at the beginning of the film and giving a small sub-atmospheric loop at the end of the film.

The Extent of the Friction-Generating Film

Conditions outside the load-carrying film are less clearly defined. In the case of the partial load-carrying film (0 to π or 0 to $\overline{\pi+\alpha}$) solutions, the remainder of the oil film is said to be cavitated and at or near atmospheric pressure, but it is assumed to contribute to the total friction an amount corresponding to a continuous film under simple shear.

It seems likely that this is an over-estimate, and that the true friction lies between that for the load-carrying film alone and that calculated on the basis just described. This possible variation is by no means small, but it tends to be obscured by the conventional presentation of friction loss as friction coefficient. For the present purpose it is better to express friction torque as a fraction of the friction torque with concentric operation and a completely continuous film under simple shear (the so-called Petroff torque).

Re-examining the theoretical results of Sommerfeld, Cameron and Wood, and DuBois and Ocvirk on this basis, the journal friction torque ratios with the minimum and maximum simple shear contributions from the unloaded cavitated part of the film have been evaluated (for width/diameter ratio unity) in Fig. 2. The



Width/diameter ratio unity
——— Sommerfeld 0 to 2π.
– – – – – Cameron and Wood 0 to 2π.
— — — — DuBois and Ocvirk 0 to 2π.
— · — · — DuBois and Ocvirk 0 to π.
— · · — · · — Cameron and Wood 0 to $\overline{\pi+\alpha}$.

Fig. 2—Friction ratio and film extent

difference between these extremes is large, and corresponds to approximately ±25 per cent of the mean. That it is seldom considered probably arises from the use of friction coefficient, where increasing load at higher eccentricities masks changes in the friction itself, and from the general uncertainty in lubrication data resulting from the need to choose a representative film viscosity. Blok (1949) has pointed out that this choice is unlikely to give good agreement between experiment and constant viscosity theory for both load-criterion/eccentricity and friction-criterion/eccentricity, and has demonstrated this in the case of Rumpf's data, where the friction was only about half of the theoretical. As shown in Table I, the friction results of Clayton and Jakeman (1936) and of Fogg (1948) are also less than Cameron and Wood's estimate.

Side-Leakage Oil Flow

The oil flow from a bearing running with a positive oil-supply pressure is the result partly of hydrodynamic pumping action and partly of the supply pressure. A complete theoretical analysis should derive the flow by considering the general pressure distribution dueto a hydrodynamic film with a pressure source of known magnitude and geometry. This has been attempted by Muskat and Morgan (1938) but their assumption of a continuous film limits the application of their results. Other analyses neglect the effect of supply pressure.

The side-leakage flow is the difference between the circumferential flows at the beginning and end of the pressure film, and may be derived from the general expression for the circumferential flow at any position θ in a bearing :

$$Q_\theta = \frac{Ubh}{2} - \frac{bh^3}{6d\eta}\left(\frac{\overline{\partial p}}{\partial \theta}\right) \quad . \quad . \quad . \quad . \quad (2)$$

where $\left(\frac{\overline{\partial p}}{\partial \theta}\right)$ is the mean circumferential pressure gradient at θ and h is assumed to be constant over the width.

The simplest assumption is that the second term is negligible at the film ends, so that the radial velocity distribution is linear. For a film extent 0 to π the side leakage Q is then

$$Q = Q_0 - Q_\pi = \frac{Ub}{2}(h_0 - h_\pi)$$
$$= \frac{Ub}{2} \cdot 2e \text{ (see Fig. 1).}$$

This can be rearranged to

$$\frac{Q}{\pi bcdN} = \varepsilon \quad . \quad . \quad . \quad . \quad . \quad (3)$$

a result obtained by DuBois and Ocvirk. $Q/\pi bcdN$ is the flow per revolution per unit clearance volume and was described by Barnard (1926) as the pumping efficiency of the bearing.

If the film starts later than the position of maximum film thickness it is obvious that the flow is reduced. For example, if the film starts at $\theta=\pi/2$,

$$\frac{Q}{\pi bcdN} = \frac{\varepsilon}{2}$$

However, the mean pressure gradient at the film beginning is not zero, especially if the bearing is wide or the film is late in starting, so the second term cannot be neglected and this also reduces the side leakage. Cameron and Wood (1949) take this into account and tabulate circumferential flows in terms of angular position for various width/diameter and eccentricity ratios. Values of the pumping efficiency $Q/\pi bcdN$ can be deduced, and these are found to be proportional to ε, the empirical expression

$$\frac{Q}{\pi bcdN} = \left(1{\cdot}056 - 0{\cdot}254\frac{b}{d}\right)\varepsilon \quad . \quad . \quad . \quad (4)$$

representing the data to an accuracy of 1 per cent for width/diameter ratios between $\frac{1}{4}$ and $1\frac{1}{4}$.

Again the expression refers to a pressure film beginning at the position of maximum film thickness $\theta=0$, and the flow will be reduced by a later start. There appears to be no analysis analogous to that of Cameron and Wood which takes this factor into account but an indication may be obtained from the partial bearing analyses of Needs (1934) and Sassenfeld and Walther (1954). The effect is shown for a bearing of width/diameter ratio unity in Fig. 3.

It is evident from these remarks that film extent has an appreciable effect on side-leakage flow, even without consideration of the lubricant supply pressure. As supply conditions might be expected to affect film extent there is a need for a more comprehensive analysis, probably on the lines of the Cameron and Wood relaxation analysis, taking into account supply pressure and film starting position.

Visual Studies of Film Extent

These considerations reinforced the view that a visual study of film extent would be worth while. Skinner (1905), Hamilton Gibson (1919), Hyde (1920), Barnard (1926), Vogelpohl (1937), Kolano (1953), and Banks and Mill (1954), have previously made observations with transparent bearings. Only Barnard, Vogelpohl and Kolano worked with complete journal bearings and in each case only a limited and qualitative use was made of the method, using very light loads, low speeds and large clearance ratios compared with practice. The experiments now described use an improved technique and cover a wider range of operating conditions.

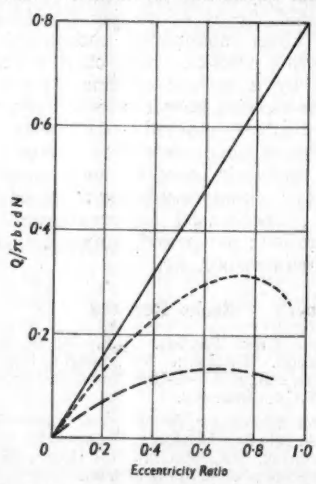
Experimental Technique for Visual Observation.—The first tests were made with a Perspex bearing bush running on a steel shaft with a clearance ratio of 0·002, and it was found that the extent of the thin oil film could be clearly defined by causing it to fluoresce under ultra-violet irradiation, film breakdown appearing as dark areas.

Unfortunately, although convenient to make, Perspex bushes with standard values of the clearance ratio (0·001 and 0·002) were unsatisfactory in operation at other than very light loads and low speeds. High loads and appreciable heat generation caused distortion and seizure.

It was then found that satisfactory results could be achieved with "Hysil" glass bushes made from "Veridia" precision-bore glass tube which is available in long lengths with ±0·0004in tolerance on bore diameter. Bushes of various forms were made from selected pieces of this material, 25mm bore by 2mm thick, and loads as high as 200 lb have been carried without bearing failure.

The outside diameter of the tube was more variable, and a method of mounting the bush so as to apply load and to measure friction and film thickness without obscuring the view has presented difficulties. In the present experiments, in which friction and film thickness data were not obtained, either a Perspex loading shoe or two stranded cables, 0·025in in diameter, were used to apply the load. To minimise the tendency for the outflowing oil to obscure the outer surface of the bush the load was applied upwards.

The particular oil used in this work, a light mineral oil Velocite E, was found to fluoresce brightly under ultra-violet light, and tests showed that the addition of a fluorescent dye was of no benefit. Other oils, such as B.P. paraffin oil, would require a fluorescent additive. The light source was originally a 1kW high-pressure mercury-discharge lamp fitted with a reflector and Chance OX7 filters, emitting principally at 3650Å. Situated approximately 15in from the bearing, this gave adequate exposures on Ilford HP3 film using $^1/_{20}$ second at f6·3 with a Wratten 2B ultra-violet absorbing



Effect of position θ of beginning of film ; width/diameter = 1.
——— Cameron and Wood, $\theta=0$.
– – – – – Sassenfeld and Walther, $\theta=90-\psi$.
— — — — Needs, $\theta=120-\psi$.

Fig. 3—Theoretical side leakage

Table I

Sommerfeld number S	Theoretical eccentricity ratio (Cameron and Wood) $\left(\frac{\text{width}}{\text{diameter}}=1\right)$	Friction torque on journal/Petroff torque					
		Cameron and Wood $\left(\frac{\text{width}}{\text{diameter}}=1\right)$	Clayton and Jakeman (180 lb per sq in load)				Fogg
			Clearance ratio 0·00075	Clearance ratio 0·00175	Clearance ratio 0·0038	Clearance ratio 0·00805	Clearance ratio 0·001
1·27	0·1	1	0·78	0·76 (extrapolated)	—	—	0·42
0·628	0·2	1·03	0·84	0·81	—	—	0·42
0·180	0·5	1·21	1·05	0·98	0·99	—	0·66
0·0452	0·8	1·93	2·69	1·6	1·83	1·5	1·57

The experimental figures are for bearings of width/diameter ratio $1\frac{1}{4}$ with a single-hole oil entry. The journal friction was deduced from the measured bush friction using experimentally determined eccentricity figures in the case of Clayton and Jakeman, and Cameron and Wood's theoretical figures in the case of Fogg.

filter on the camera lens, but ½ second is to be preferred for a good negative. A cine-film has been made at eight frames per second, corresponding to 1/15 second at f4, using Kodak Super XX reversal film and two lamps. Latterly, electronic flash has been used, with 100-joule xenon tubes and the same filters at approximately 6in from the bearing, the lens aperture being set at f9.

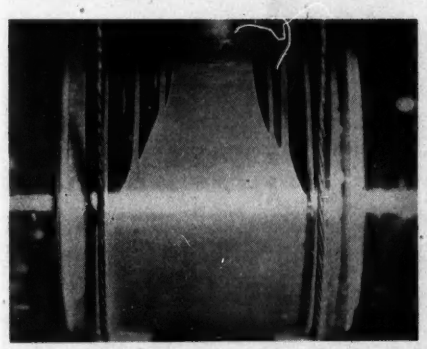
Both sides of the glass bearing were photographed at the same time using two Leica 35mm cameras, with 5cm lenses to give 1/1 ratio.

Speed, 1500 r.p.m.; load, 10 lb per square inch; supply pressure, 6 lb per square inch; width/diameter, 1·1; clearance ratio, 0·0012.

Fig. 4—Ruptured inlet film (S=3)

To evaluate the angular extent of the oil film on the photographs, a scale was prepared by photographing an engraved shaft fitted inside the bush and supplied with oil. The effects of distortion due to the thickness of the glass bush were overcome by this means, but the scale was nevertheless compressed and difficult to read near the top and bottom edges of the bearing.

The test machine consisted of a horizontal ground steel shaft (diameter 0·9830in or 0·9820in average axial finish 3·6 micro-inches) carried in two plain bearings and driven by a variable-speed motor. The central floating glass bush was loaded by a cable and weights. Oil was admitted to an oil hole (0·1in in diameter) in the bearing from either a Mariotte bottle for small heads or a variable-pressure oil pump system. Oil inlet temperature and pressure were measured near the test bearing and, where necessary, a



Width/diameter, 1; clearance ratio, 0·002; load, 65 lb per square inch; speed, 1500 r.p.m.; supply pressure, 3 lb per square inch
Single oil hole.

Fig. 5—Typical inlet pattern (S=0·15)

correction was applied for the loss of head in the oil pipe. A thermo-couple was positioned at each side of the bearing so that a mean value of the temperature of the oil outflow was obtained just as it left the bearing. The flow was collected in a measuring cylinder for flow timing purposes.

Range of Experiments.—The results described here are for film extent and oil flow only, although methods for the simultaneous measurement of friction torque and film thicknesses with the glass bearings are being developed. The bearing sizes used were nominal bore, 0·984in; nominal width, 0·5in, 1in, 1·5in; nominal diametral clearance, 0·001in, 0·002in. The various forms of oil inlet were single oil hole, 0·1in diameter; single oil hole with axial groove, 0·12in by 0·8in; single oil hole with relief for ±45 deg., width 0·8in; single oil hole with complete circumferential groove, width 0·1in. All grooving was approximately 0·05in deep, and the machining produced an arcuate cross-section with sharp edges.

The oil inlet position was diametrally opposite the loading point, although some tests were made on a single oil hole at both 90 deg. in advance and retard of this position.

The operating conditions were speeds, 500–5000 r.p.m.; loads, 10–100 lb per square inch; oil supply pressure, 0–26 lb per square inch; lubricant, Velocite E (see Table II for viscosity).

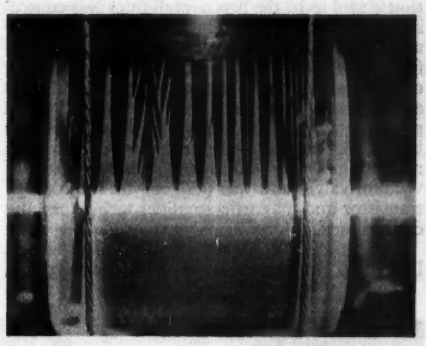
TABLE II—*Viscosity of Velocite E*

Temperature, deg. Fah.	50	75	100	125	150
Viscosity, centistokes	31·6	17·2	10·8	7·3	5·2

QUALITATIVE DESCRIPTION OF FILM CONDITIONS FOR SINGLE OIL HOLE BEARINGS

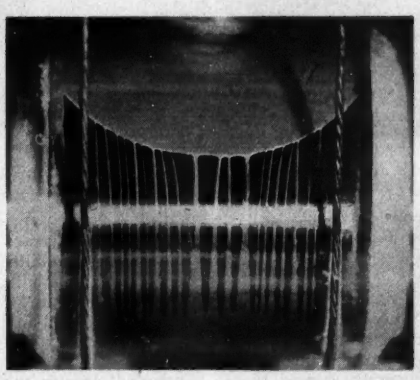
Under very light loads (S~10) the bearing operates at low eccentricities and the film is complete, particularly at lower speeds. At higher loads (S~4) the inlet film only may rupture and form cavities which are surrounded by a complete film (Fig. 4). These are not altogether fixed in position and shape, and sometimes form as a result of air passing down the supply line and remaining obstinately in the inlet film.

On increasing the load further (S~0·6) the film ruptures more completely, and the characteristic single hole inlet pattern forms (Fig. 5). The oil issuing from the inlet hole spreads and forms a full-width film after travelling some distance round the bearing. This full-width film passes through the loaded region and on the outlet side a more or less linear breakdown occurs (Fig. 6), although the breakdown line may be inclined to the shaft axis. Theory (Cameron and Wood, 1949, Sassenfeld and Walther, 1954) suggests that the breakdown profile should be convex owing to side leakage effects. The form of breakdown is characteristic of all the forms of bearing examined, consisting of a number of



Width/diameter, 1; clearance ratio, 0·002; load, 105 lb per square inch; speed, 1500 r.p.m.; supply pressure, 3 lb per square inch

Fig. 6—Typical outlet film breakdown (S=0·1)



Width/diameter, 1·1; clearance ratio, 0·0028; load, 100 lb per square inch; speed, 500 r.p.m.; supply pressure, 25 lb per square inch

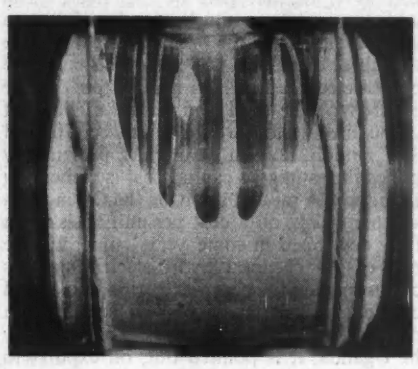
Fig. 7—Outlet film showing backward spread of inlet film (S=0·02)

thin oil filaments which traverse the remaining part of the bearing relatively undisturbed and rejoin the inlet film. The filaments are finer and more closely spaced at the higher eccentricity ratios. There is no foaming at the point of breakdown, as has been surmised by some writers.

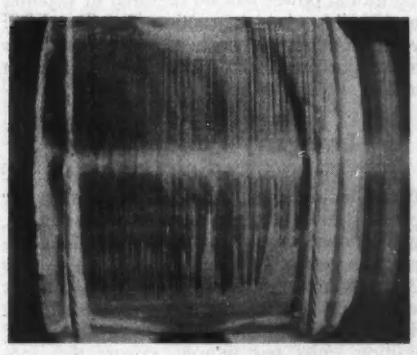
In the lower part of the speed range investigated the film pattern is remarkably steady. At higher speeds the outlet filaments fluctuate laterally, and oil creeps back along the inner surface of the bush on the outlet film side of the oil hole, to be carried forward again by the shaft in an irregular closed circulation.

Increase of the oil supply pressure has little effect on the angular position of the outlet film breakdown. However, the spreading inlet film is broadened considerably and attains full width somewhat earlier, probably in order to discharge the extra oil flow induced by the higher supply pressure. At low speeds and high supply pressures the inlet film spreads backward into the outlet side of the bearing, especially with large clearances (Fig. 7).

Even at very small supply pressures, as low as a few inches head of oil, the full-width film extent is maintained as above, while on shutting off the oil supply altogether the inlet film pattern merely inverts (Fig. 8*a*) and oil is supplied from the



a—Oil supply from meniscus.



b—Oil supply from meniscus failed.

Speed, 1250 r.p.m.; load, 40 lb per square inch; supply pressure, 0 lb per square inch

Fig. 8—Inlet film pattern with oil starvation

meniscus at each side of the bearing, the side leakage oil recirculating to the meniscus. If the oil meniscus fails the circumferential extent of the film is appreciably reduced (Fig. 8*b*), but hydrodynamic conditions may be maintained for some time. A drip feed produces an alternation between the normal and inverted inlet film shapes. The influence of the meniscus is evident even in the normal inlet pattern: the full width film forms as the spreading inlet stream joins the inflow from the meniscus.

Some tests were performed with oil-filled chambers at the bearing sides to see whether virtual immersion in an oil bath modified the film extent in the low-pressure regions of the bearing oil film. There seemed to be little difference, and observation of the process of film rupture suggested that it resulted from air (or oil vapour) within the film rather than from air

admission from the bearing sides. Usually, a complete oil meniscus is present at the bearing sides and it appears to function as a seal as already recounted.

Tests with couples applied in addition to the load to simulate misalignment showed that large couples in the plane of the load and bearing axis were necessary before appreciable film pattern asymmetry resulted. However, twisting couples in a plane perpendicular to the load produced a considerable effect.

Bearing whirl—frequency approximately half shaft speed—has been observed at light loads, and it appears to be associated only with the complete circumferential film described above.

(To be continued)

Hydro-Electric Surveys in Uganda

Surveys of the Victoria Nile have recently been completed for the Uganda Electricity Board by the consulting engineers, Sir Alexander Gibb and Partners and Messrs. Kennedy and Donkin. There is an additional potential output of some 500 MW in the nineteen-mile stretch of river below the existing Owen Falls hydro-electric scheme, it is considered, thus providing a total capacity of 650 MW, in that part of the river. The additional power could be developed in two or three stages but the present indications are that three dams and power stations could be constructed, the first having an output of over 150 MW. The total firm output of the first of the three stations would be between 700 and 800 million units a year. At present-day prices the estimated capital cost is rather less than the £16 million spent on the Owen Falls scheme which, on a cost per unit basis when fully developed in some years' time, will be one of the cheapest hydro-electric stations constructed in the post-war period, the Board considers. The start of new construction depends on the growth of the use of electricity in Uganda, it is pointed out, on exports to Kenya, and also on railway electrification, but on present estimates it seems likely that additional power will be required in Uganda by 1964/65. As construction takes about five years, preliminary work may therefore have to start before 1960. A detailed examination of one site is now being made.

Invergarry Power Station

The North of Scotland Hydro-Electric Board has brought into commission Invergarry power station in Inverness-shire. This station, will produce about 82 million units of electricity in a year of average rainfall, from a single 20 MW turbo alternator set, and is the lower station of the Board's Glen Garry scheme. This scheme, which will produce about 168 million units per annum, covers the construction of a dam at the outlet of Loch Quoich, a power station on the Gearr Garry river and another dam at the outlet of Loch Garry from which water is carried by tunnel to the Invergarry power station. The Quoich power station began to operate in December. The Invergarry power station which is alongside the main Fort William–Inverness road, is mostly underground, only the top of the stone building being visible. The main generating set is powered by a Kaplan turbine operating under a head of 175 feet.

British Standards Institution

All British Standard Specifications can be obtained from the Sales Department of the Institution at 2, Park Street, London, W.1

RACK TYPE GEAR CUTTERS

No. 2697 : 1956. Price 2s. 6d.—This new British Standard includes tables of dimensions and permissible errors in rack type gear cutters. It indicates the method of mounting and shows the tooth form and explains with the use of diagrams the terms used. Two appendices give details of the marking and instructions to be used when ordering cutters.

Automatic Thickness Control on Five-Stand Tandem Cold Reduction Mill

BY OUR AMERICAN EDITOR

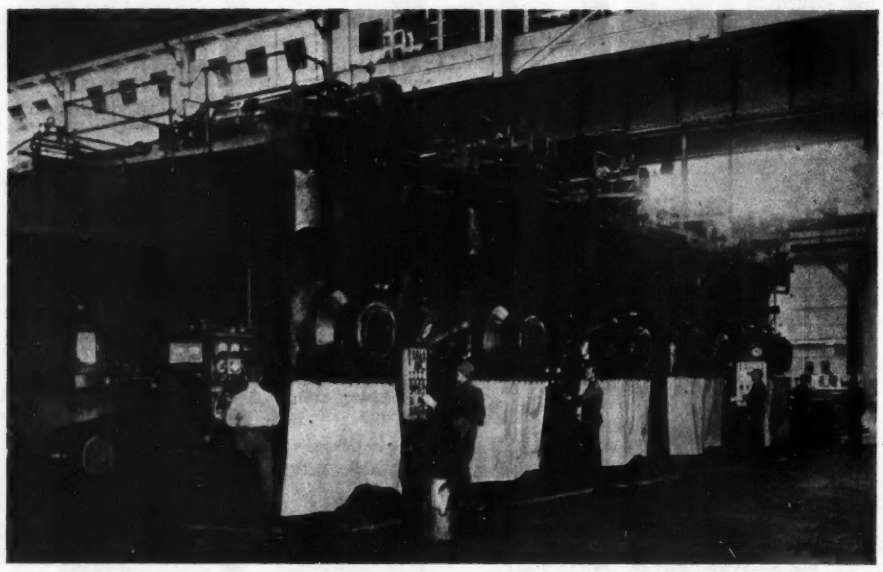
AN interesting X-ray gauge system applied to obtain closer, continuous control of cold reduction mills producing sheet and strip products has been developed through the co-operative efforts of the United States Steel Corporation and the General Electric Company, Schenectady, New York. After adequate tests and experiment, this system is now operating on a 21½in and 53in by 43in four-high, five-stand tandem cold reduction mill at the U.S. Steel Irvin Works at Dravosburg, near Pittsburgh, Pennsylvania.

Since the advent of the high-speed continuous sheet and strip mills in the steel industry, the need for automatic thickness control has become more and more pressing. To solve this problem, a joint research and development study of cold reduction mill controls was instituted by the United States Steel Corporation and the General Electric Company. Before attempting to design any system, engineers of General Electric submitted the problem to an electronic differential analyser for a study of the complete cold reduction mill thickness problem and drive system. The analysis took into account not only all electrical constants, but changes in roll pressure and inter-stand tension. The resulting installation is said to be the first overall thickness control in the American steel industry to utilise not only a screwdown control to vary reduction on the first stand, but also a vernier system operating on the roll speed of the last stand. With this arrangement, thickness variations in the finished product can be held smaller than was possible by manual means. An interesting aspect of the regulating system is that it is completely free of vacuum valves and was designed on the basis of transistor and magnetic-amplifier techniques and components.

The tandem cold reduction mill, which is illustrated in the accompanying photograph, consists of five roll stands and a tension reel arranged in a continuous line, so that strip passes successively through the roll stands and is coiled under tension as delivered from the last stand. For several years it has been customary to use an X-ray thickness gauge between the last stand and the winding reel to obtain a continuous indication of the thickness of the finished strip. It is generally agreed that the principal reason for variation in the finished thickness of strip is because the incoming strip has variations in thickness. It has also been found by experiment that these variations can be substantially reduced through control of the roll opening and roll speed at stand 1. Further cold reducing renders the steel strip more brittle and susceptible to breakage, thus reducing the permissible range of control in later stands. Thus even in the case of a manually controlled mill, an additional gauge might be suggested between stand 1 and 2. The problem then was to develop a means of automatically operating controls to obtain a more uniform finished strip thickness.

In this particular case, the solution was to use electrical feedback control or regulating systems. These electrical systems continuously monitor performance. They do not suffer from operator fatigue and, with electrical amplification, improve accuracy. Because they also have considerable electrical power they can speed up response to make corrections faster. The automatic gauge control system as developed consists of two portions. A coarse system utilises an X-ray gauge between stands 1 and 2 and a regulating system to operate the screw-down motors on stand 1. A vernier system utilises the X-ray gauge after stand 5 and a regulating system to vary stand 5 drive motor speed to effect gauge control. The complete gauge and drive system was designed on the basis of extensive analytical and electronic computer studies. The analyses took into account not only all electrical constants, but mechanical constants involving roll pressure and inter-stand tension as well as the inherent ability of the basic drive system to perform minor "ironing out" action of variations in incoming gauge.

In the coarse system an X-ray gauge is mounted between stands 1 and 2 to measure the thickness of strip leaving stand 1. The X-ray gauge is built in such a way that the desired thickness is set on the X-ray gauge as a reference signal. The difference between actual thickness and desired thickness is brought out of the gauge as a deviation or error signal. This signal is fed into a transistor amplifier system which determines the direction of the error, i.e. whether the strip is heavy or light. This polarised signal is then fed into an "amplistat" automatic control system. This automatic control system is designed so that if the strip coming out of stand 1 is 0·0005in off gauge, the screwdown motors will run for a shorter time than if the strip were 0·001in off gauge. This method of operation requires the screwdown motors to jog less frequently to bring the thickness back into tolerance than if the timed pulses or "on" times were always equal. A check interval is



Five-stand tandem cold reduction mill at the Irvin works of the United States Steel Corporation

provided between timed pulses to permit the strip to get from the rolls of stand 1 to the X-ray gauge position before the automatic control

Indicating head of X-ray thickness gauge mounted at operator's position on five-stand tandem mill

makes any further correction on the screwdown motors. The check time or interval between timed pulses is also a variable since the strip speed varies and the distance is fixed between the stand rolls and the X-ray gauge.

Several auxiliary functions are incorporated in this coarse regulator, such as automatic "screw

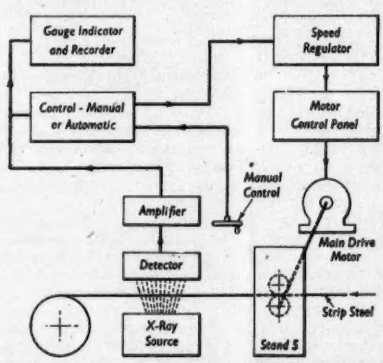


Diagram of vernier gauge system incorporated in stand 5 drive of five-stand tandem mill to obtain automatic thickness control

reset," "screw departure indication" and "screw departure limits." Most of the off-gauge in the material supplied to a cold reduction mill occurs at the beginning and the end of the hot bands making up the coil being rolled. The operator will select a normal screw opening which will give him "on-gauge" tolerance during mill acceleration. The automatic gauge control will maintain the desired thickness over the entire strip and at the end of the coil any deviation of the screwdowns from the desired entering thickness will be automatically removed by the "screw reset" circuit which will run the screwdowns back to the original setting made by the operator. The "screw departure indicator" is a meter which indicates to the operator how far the screws have departed from his original selected opening. There will be times when the automatic gauge control will want to run the rolls either too far apart or too close together which might produce improper mill operation. If the screws open too wide slipping will occur. If the rolls come too close together, the stand motor might become overloaded and shut down the mill. The "screw departure limit" circuit looks for such misoperation and it disconnects the automatic control when preselected limits are reached. At the same time an alarm light is turned on to tell the operator that his control is off automatic operation. Another auxiliary function incorporated in this control is automatic "regulator inspection." This function is set up in such a manner that if the regulator does not reduce thickness error during a preselected time limit, the regulator is removed from operation. Again, the operator is notified by a signal light.

The stand 1 or coarse gauge regulating system is designed to reduce large thickness variations to small thickness variations. These small thickness variations are further reduced by the operation of the vernier regulator on stand 5. The vernier gauge system is integrated into and works through feedback control systems associated with the last stand drive system. The thickness of the finished strip can be controlled over a limited range by varying stand 5 speed and the tension between stands 4 and 5. The vernier automatic gauge control system uses this principle to control the final gauge of the steel strip. An X-ray gauge similar to the one used between stands 1 and 2 measures thickness of the strip coming out of stand 5. Here, again, the desired thickness is set on the X-ray gauge as a reference signal. The gauge measures the actual thickness and the difference between the two is fed into the automatic gauge control circuits as a deviation or error signal. A transistor system determines the polarity of the signal and amplifies it to a power level suitable to be fed into the motor speed regulating system. Stability networks are incorporated in the transistor amplifier circuits to stabilise the operation of the regulator when controlling gauge.

The vernier system also has auxiliary functions incorporated. Since the regulator must necessarily have a limited range to prevent looping and breakage, some provision must be made to tell the operator when the regulator is out of range so that he may make the necessary manual adjustment of either the screw setting or the coarse or vernier speed control. This function is provided in two ways, one being by means of a regulator "operating range indicator," which is mounted on the operator's cabinet. As long as the meter indicates within range, the operator has no particular problem. The automatic control will regulate the gauge automatically. However, if the meter indicates to one end of the scale and stays there, the regulator limit indicator will light a signal to tell the operator that the regulator is out of range and that manual adjustments must be made on the stand 5 control. Regulating system inspection functions as described for the coarse regulator are included to protect against malfunctioning of the system components.

The "Hochschule Für Gestaltung" at Ulm

The Hochschule für Gestaltung of Ulm, German Federal Republic, the official opening of which took place at the beginning of last October, is in many respects a remarkable enterprise. This residential college, as its name states, is a university for design, covering in its activities the whole range of applied art from architecture to kinematography and the printed and spoken word. Students are accepted on the basis of ability and previously acquired technical and professional knowledge in their chosen field. The diploma courses, including basic training, are arranged to last four years, and the greatest stress is laid at all stages upon a close relation between theoretical understanding and practical work.

In its origins, the Hochschule für Gestaltung goes back to the years after the war, when the need for a higher centre for design was realised by Frau Inge Aicher-Scholl, founder of the Volkshochschule at Ulm, and her husband, Otl Aicher. A first attempt to set up such an institute on a small scale failed, owing to the devaluation of the currency in 1948. The idea was revived in 1949, when first the Norwegians, and later the United States High Commission under Mr. McCloy, offered assistance. It was at this time that Frau Aicher-Scholl and her husband were joined by the architect Max Bill, who is now rector of the college. After protracted negotiations, a written preliminary decision was received from the High Commission for Germany at the end of 1950, offering a gift of DM.1,000,000 from the McCloy Fund, on condition that a sum of DM.700,000 was raised from German sources within eight months, and that the state undertook to make good any deficit for three years. At the same time, legal recognition was given by the state of Baden-Württemberg to the Scholl Foundation, the parent institution of the proposed college. Negotiations had already begun with the Ministry

Fig. 1—View of the buildings on the Oberer Kuhberg near Ulm, seen from the South.

Fig. 2— The central hall between the administration offices and the studios and lecture rooms. Internal walls are of bare concrete and brick.

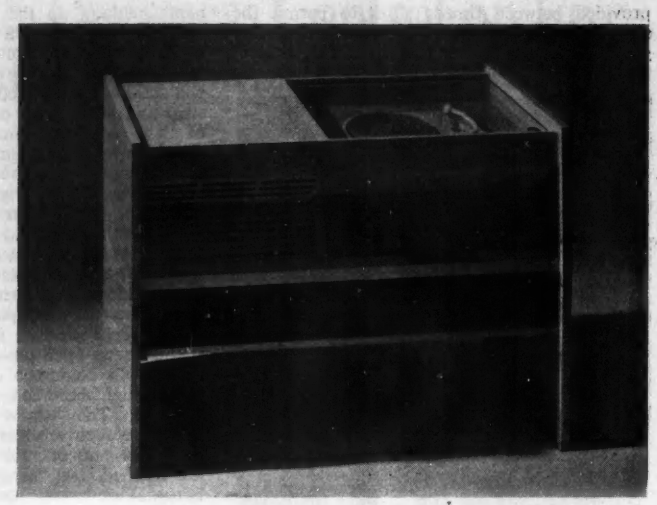


Fig. 3— Radiogram cabinet. An example of the work of the department for Product Design.

of Finance in Bonn regarding the site. Of paramount importance, however, was the recognition of the Scholl Foundation on a charitable undertaking so that any contributions would carry exemption from tax. This was obtained at the beginning of June, 1951, leaving three months in which to raise the amount required by the McCloy Fund. It was of great help to the organisers at this critical period that they were joined by a man with legal training, Mr. Hellmut Becker, the son of the late Prussian Minister for culture, C. H. Becker. A particular stumbling block was the very novelty of their conception, which led, perhaps naturally, to an increased resistance on the part of the authorities, who, on the whole, were most reluctant to give any help to which the tightest administrative strings had not been attached. By contrast, the cultural affairs committee of Baden-Württemberg decided unanimously to subsidise the college provided its independence were guaranteed, stating that in the committee's opinion new developments could only succeed if private initiative were given a free hand. In July, 1951, the town council of Ulm declared its willingness to make a foundation contribution and to give aid by the subsidies in kind, in the form of fuel, gas and electricity. The guarantee of governmental support required by the Americans had meanwhile been obtained, and the sum of German contributions was finally raised in time. At that juncture, malicious allegations of Communism were brought against members of the Scholl family by a former Gestapo agent—Hans and Sophie Scholl, after whom the Foundation is named, had been executed in 1943 as members of the Weisse Rose resistance group at Munich University—and those denunciations led to the scheme being suspended for nearly a year while the Federal Government and the High Commission carried out investigations. Eventually, the matter was cleared up, and the American funds were handed to the Foundation in June, 1952.

Actual construction began in September, 1953. Plans had been drawn up by Max Bill in 1950, but had had to be modified as the scope of the college had increased. Detail design was undertaken by a team of students of architecture, led by architect Fritz Pfeil. One of the difficulties of this part of the work was caused by the fact that much of the assistance given by industry consisted of supplies of materials to which the design had to be adapted. Our illustration (Fig. 1) shows the group of buildings on the site of the Oberer Kuhberg, south-west of Ulm. From left to right are seen the studios and workshops of the departments of product design and visual communication, next the two-storey block of the departments of basic training, architecture and town planning, followed by the administration and library building (behind trees), and the double-storey building housing the lecture hall, dining hall, and kitchen. In the right foreground is the porter's lodge, while beyond the trees may be seen the first of three blocks of students' living quarters. Houses for the staff are grouped to the south-west.

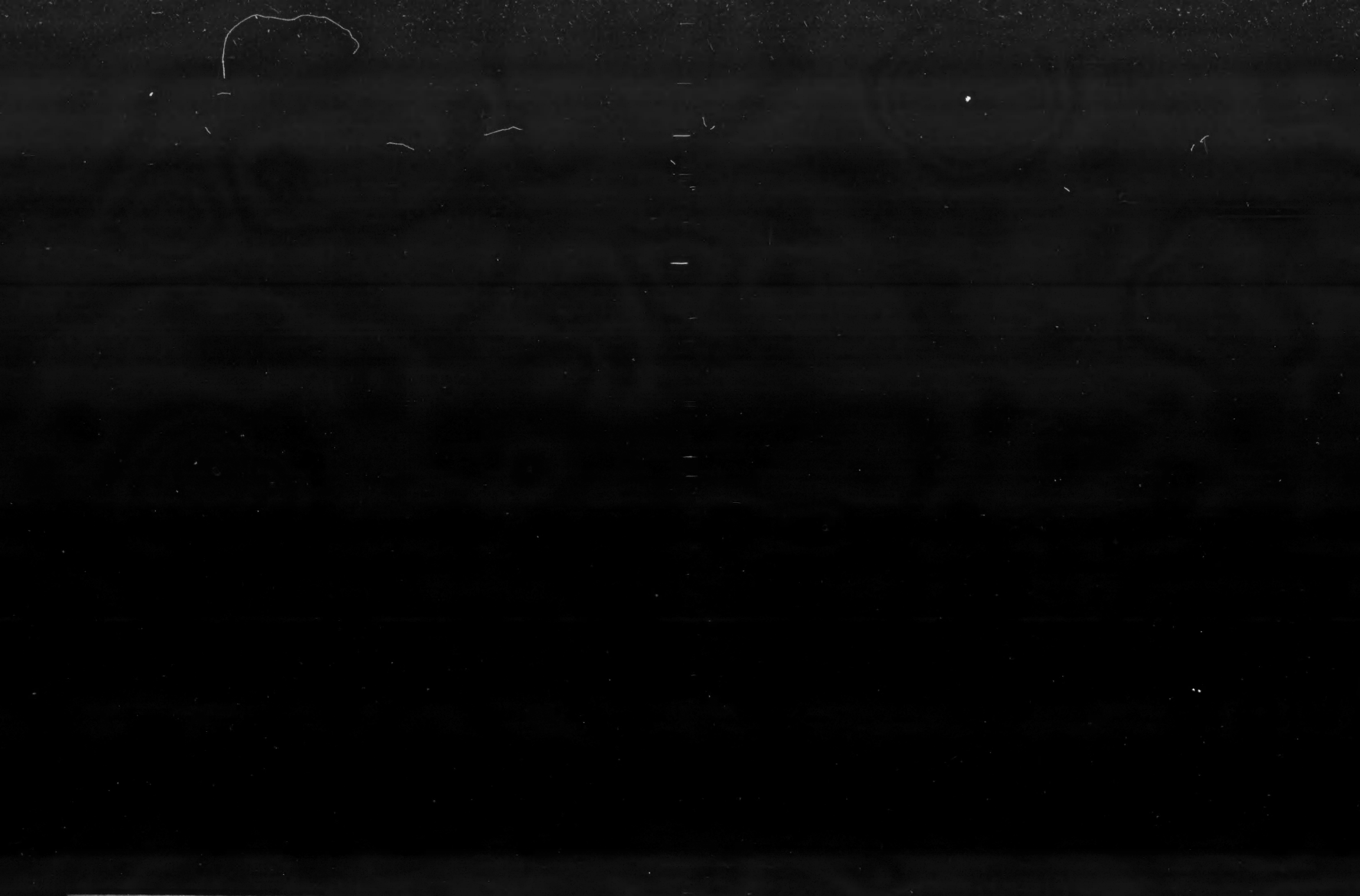
With the exception of the five-storey living blocks, all the buildings are either single or double-storey. They are constructed in reinforced concrete ; all load-bearing components and outside walls, as well as many internal walls, are left with bare concrete surfaces (Fig. 2). The design is based throughout on the same material and on a single module.

Arranged in five groups among existing and newly planted trees near the hilltop, the buildings are set out on the southern slope in such a way that no concentration of volume is created anywhere in the area. In the centre is a garden and terrace, and several courtyards with gardens serve to break up the broad single-storey block of the departments of product design and visual communication at the top of the hill, which is accessible by road from the north.

Fig. 4— Example of door handle; another instance of the work carried out by Product Design.

Although the first workshops were brought into use as early as August, 1954, and the whole of the teaching activities was transferred to the new building at the beginning of last year, many portions of the interior still remain to be completed, a task which is being undertaken by students of product design as part of their training.

The Hochschule für Gestaltung is an association of independent and complementary departments, viz. those for product design, architecture, town planning, information and visual communication, in addition to which there is the department for basic training. Students work in groups in close contact with their teachers on the solution of practical problems, and are taught the required theory in connection with those problems.

A number of institutes for research in special fields will be associated with the Hochschule. One of these, the Institute for Product Design, has already commenced work on its own and sponsored industrial research and design (Figs. 3 and 4). The two examples may serve to illustrate the aim which the college has set itself : to help in the creation of a broad culture of peoples' daily existence, a prerequisite for which is the taking of infinite pains over innumerable details such as those shown here.

GAS TURBINE DEVELOPMENTS.—The Winter, 1956, issue of the *Bristol* Quarterly gives an account of a marine version of the " Proteus " free-turbine shaft-power engine. Basically a " Proteus 755," the marine engine has all magnesium zirconium parts replaced by equivalents in aluminium alloys : on test, sea water was sprayed into the air intake at twice the rate hitherto experienced in service for 225 hours to test for corrosion fatigue salt deposits on the compressor blades being removed by injecting distilled water with the engine idling. Aircraft accessories such as constant speed unit, parking brake, combined control unit linkage and 50kVA alternator, have been replaced by marine components such as overspeed fuel governor, overspeed trip equal to sudden unloading at full power and inching motors which rotate the compressor and turbine rotors at 10 r.p.m. to prevent brinelling of the bearings. In place of the airscrew and reduction gear there is a Layrub shaft and an Allen-Stoeckicht epicyclic gear. Since the free-turbine engine is very flexible in regard to output speed, a reverse gear is favoured rather than a variable-pitch propeller. It is pointed out that where a " Proteus " marine engine is used as a high-speed power plant with diesels for manœuvring, no clutch will be required as the power turbine will impose negligible load on the water-milling propeller. A particularly compact installation can be achieved by using a gearbox with input and output shafts inclined at a reflex angle, so that the turbine is at the aft end of the engine-room exhausting through a short duct over the stern. Due to the large area of the intake annulus, the engine-room can serve as a plenum chamber. Entirely satisfactory running for 500 hours on Admiralty gas oil has been undergone, the turbine inlet temperature has been reduced from 1600 deg. Fah. in the aircraft application to 1480 deg. Fah. in the marine version, the power output being 3500 h.p. at 11,450 r.p.m. The same issue contains an article on the testing of " Olympus " engines in an Avro " Ashton " : the wing being virtually that of the earlier propeller-driven marks of " Tudor," it proved easy to install two more jet engines in outboard pods. Two camera-monitored instrument panels are provided in the pressure cabin, since the engines can be tested independently ; it is forecast that, later on, an " Orpheus " will take the place of the starboard " Olympus." The six jets provide an initial rate of climb of 4000ft per minute and the stalling speed is 80 knots below the limiting Mach number at 40,000ft, but this is the normal maximum height for test flying. It has subsequently been announced that the " BOl. 11 " has completed a 150-hour type test at 12,000 lb thrust.